CHAPTER 16
GAS TURBINE BEARINGS AND SEALS

BEARINGS

Journal bearings provide radial support for the rotating equipment and thrust bearings provide axial positioning for them. Ball and roller bearings are used in some aircraft jet engines. However, all industrial gas turbines use journal bearings.

Journal bearings can be split or full-round. Large-size bearings used for heavy machinery normally have heavy lining. Precision insert-type bearings used commonly in internal combustion engines have a thin lining. The majority of sleeve bearings are of the split type for convenience in maintenance and replacement. Figure 16.1 provides a comparison of different types of journal bearings. The following is a description of the most common types of journal bearings:

1. *Plain journal.* The bearing is bored with equal amounts of clearance between 1.5×10^{-3} in (3.8×10^{-3} cm) and 2×10^{-3} in (5×10^{-3} cm) per inch of journal diameter between the journal (portion of the shaft inside the bearing) and the bearing.
2. *Circumferential grooved.* This bearing has the oil groove at half the bearing length. This design provides better cooling. However, it reduces the load capacity by dividing the bearing into two parts.
3. *Cylindrical bore.* This bearing type is commonly used in turbines. It has a split design. The two axial oil-feed grooves are at the split.
4. *Pressure or pressure dam.* This is a plain journal bearing with a pressure pocket in the unloaded half. The pocket has the following dimensions:

   Depth: 0.031 in (0.079 cm)

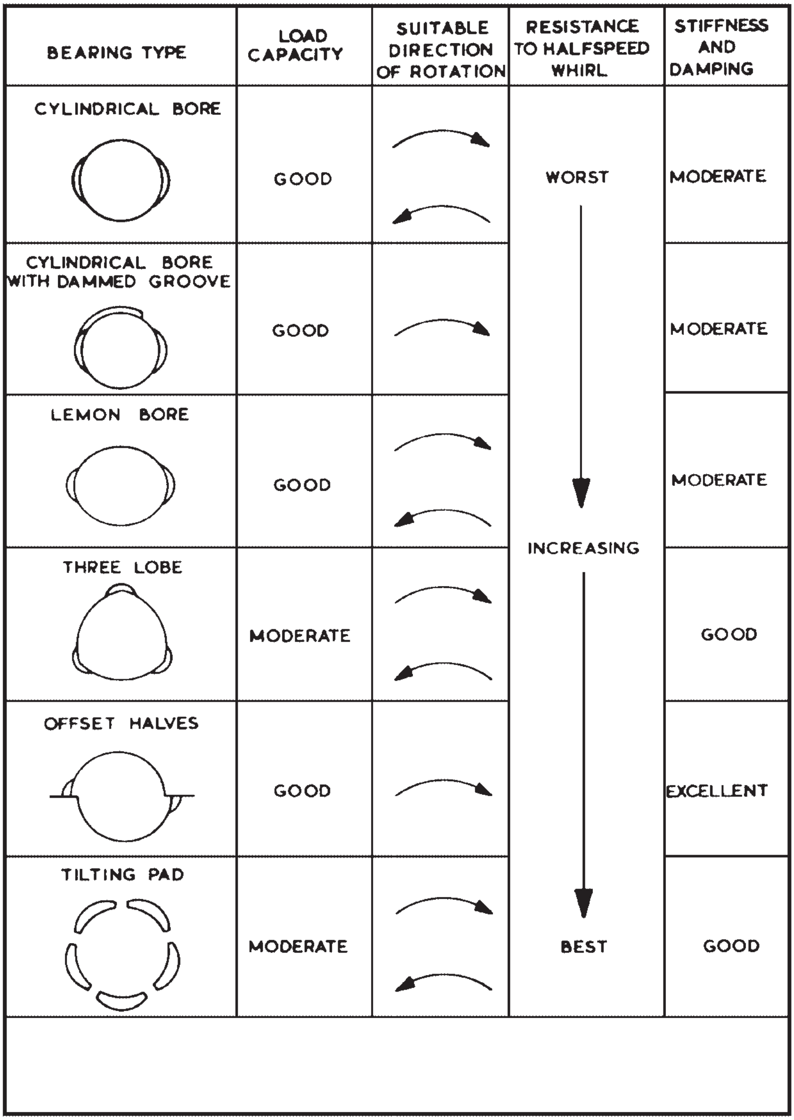
   Length: 50 percent of bearing length

   Width: arc of 135°

   The arc terminates abruptly in a sharp-edge dam. The shaft rotation is such that the oil is pumped through the channel toward the sharp edge. These bearings have only one direction of rotation. They are known to have good stability.
5. *Lemon bore or elliptical.* This is bored at the split line. The bore shape is similar to an ellipse, having a major axis approximately twice the length of the minor axis. These bearings are used in both directions of rotation.
6. *Three-lobe.* These bearings have moderate load-carrying capability and can operate in both directions. They are not commonly used in turbo machines.
7. *Offset halves.* This bearing is similar to the pressure dam bearing. It has good load-carrying capability. However, it is limited to one direction of rotation.

8. *Tilting-pad.* This is the most popular type in modern machines. It has several bearing pads located around the circumference of the shaft. These pads can tilt to assume the most effective operating position. Its main advantage is the ability for self-alignment. This bearing provides the greatest increase in fatigue life due to these advantages:

- Self-aligning to provide optimum shaft alignment.
- The backing material has good thermal conductivity. It dissipates the heat developed in the oil film



BEARING TYPE	LOAD CAPACITY	SUITABLE DIRECTION OF ROTATION	RESISTANCE TO HALFSPEED WHIRL	STIFFNESS AND DAMPING
CYLINDRICAL BORE	GOOD		WORST	MODERATE
CYLINDRICAL BORE WITH DAMMED GROOVE	GOOD			MODERATE
LEMON BORE	GOOD			MODERATE
THREE LOBE	MODERATE		INCREASING	GOOD
OFFSET HALVES	GOOD			EXCELLENT
TILTING PAD	MODERATE		BEST	GOOD

FIGURE 16.1 Comparison of general bearing types.

- The Babbitt layer is thin [around 0.005 in (0.013 cm)]. Thick babbitts reduce the bearing life significantly. Babbitt thickness around 0.01 in (0.025 cm) reduces the bearing life by more than half.
- The thickness of the oil film has a significant effect on the bearing stiffness. In tilted-pad bearings, the thickness of the oil film can be changed by the following methods:

  Changing the number of pads
  Changing the axial length of the pads
  Directing the load on or in-between the pads

These are the most common types of journal bearings. They are listed in the order of growing stability. As the stability increases, the cost and efficiency of the bearing decreases. All anti-whirl bearings impose a parasitic load on the journal. This generates higher power losses, requiring larger oil flow to cool the bearing.

BEARING DESIGN PRINCIPLES

In a journal bearing, a full film of fluid separates the stationary bushing from the rotating journal. This separation is achieved by pressurizing the fluid in the clearance space until the fluid forces balance the bearing load. The fluid must flow continuously into the bearing and maintain the pressure in the film space. Figure 16.2 illustrates the four methods of

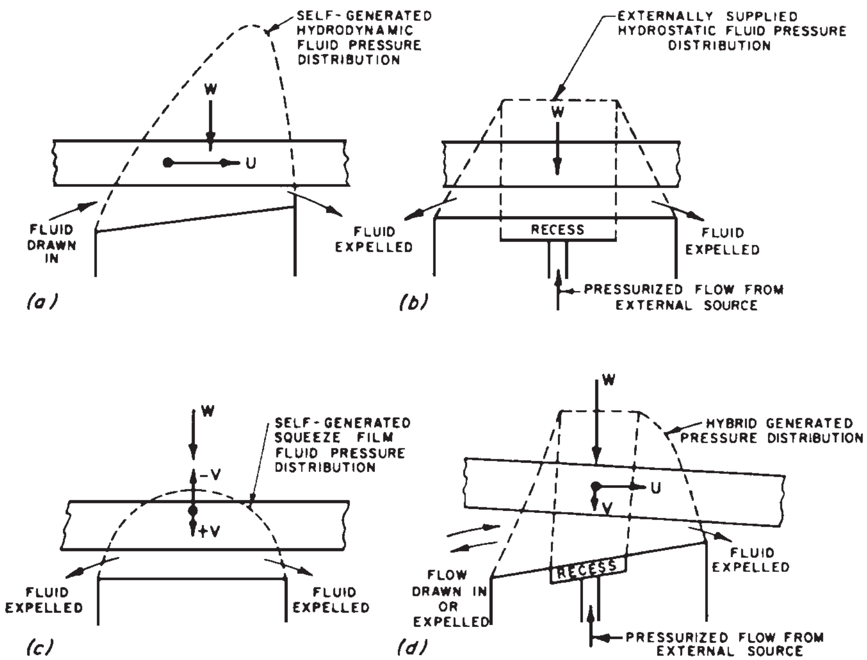


FIGURE 16.2 Modes of fluid-film lubrication: (*a*) hydrodynamic, (*b*) hydrostatic, (*c*) squeeze film, and (*d*) hybrid.

lubrication in a fluid-film bearing. The most common method is the *hydrodynamic.* It is known as a *self-acting* bearing.

Figure 16.3 illustrates the natural wedge formed by a journal bearing and the pressure distribution inside the bearing. The thickness of the fluid-film varies from 0.0001 to 0.001 in (0.00025 to 0.0025 cm) depending on the lubrication method and application. There are peaks and valley in every surface regardless of its finish. The average asperity height is around 5 to 10 times the RMS surface finish reading. When a surface is abraded, an oxide film will form on it almost immediately.

Figures 16.4(*a*), (*b*), and (*c*) illustrate three types of separation between the journal and the babbitt in a bearing:

a. Full-film

b. Mixed-film (intermediate zone)

c. Boundary lubrication

If a full-film exists, the bearing life would be almost infinite. The limitation in this case would be due to lubricant breakdown, surface erosion, and fretting of various components. Figures 16.4(*d*) and 16.4(*e*) illustrate the effect of oil additives, which are considered contaminants that form beneficial surface films.

Figure 16.5 describes the bearing health by plotting the coefficient of friction versus *ZN*/*P,* where *Z* is the lubricant viscosity in centipoises; *N,* the rpm of the journal; and *P,* the projected area unit loading. The lowest friction is reached when the full-film is

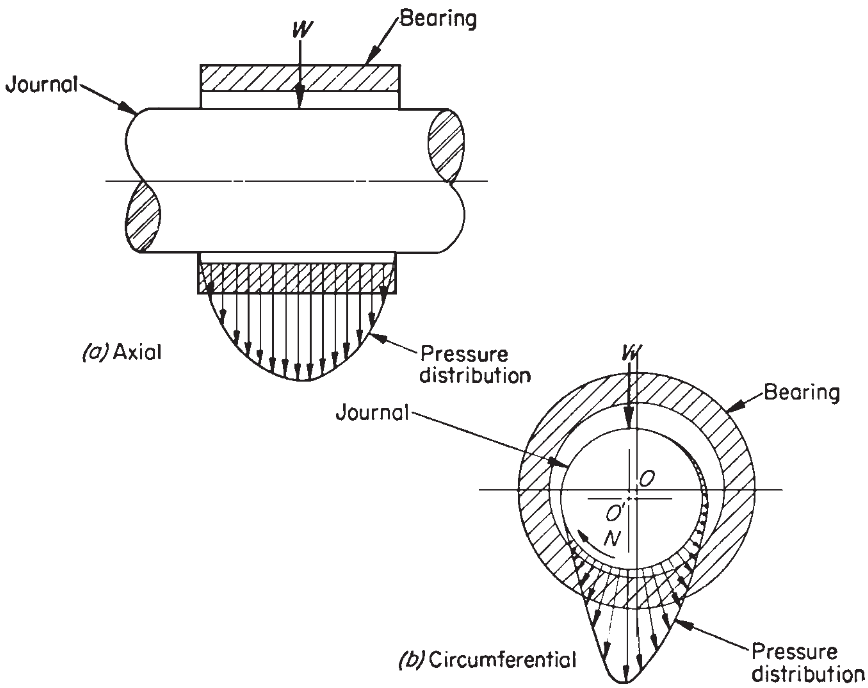


FIGURE 16.3 Pressure distribution in a full journal bearing.

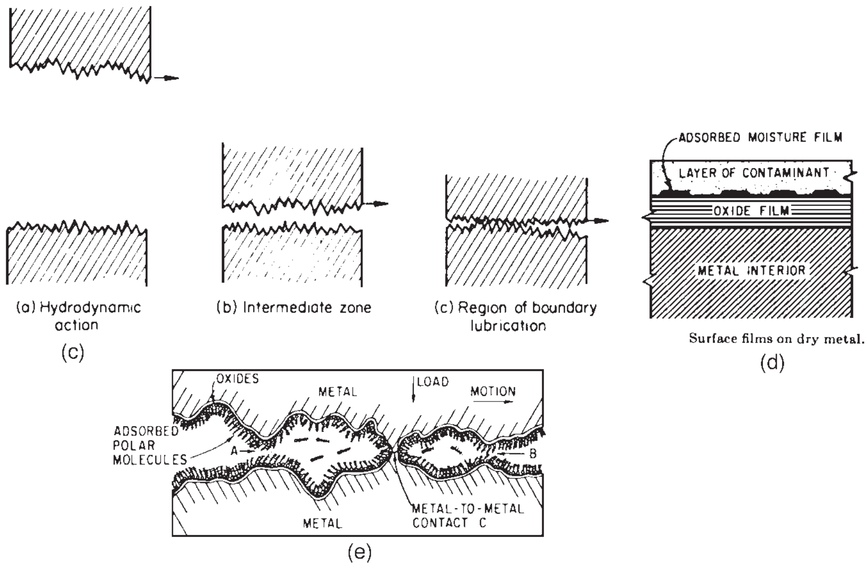


FIGURE 16.4 Enlarged views of bearing surfaces.

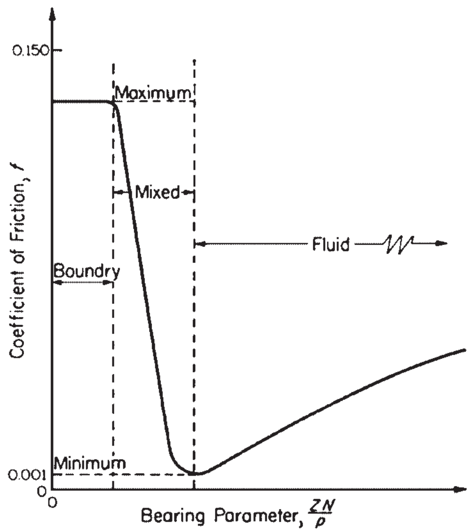


FIGURE 16.5 Classic *ZN*/*P* curve.

established. At higher speeds, the friction increases due to an increase in the shear force of the lubricant.

The transition from laminar to turbulent flow in the bearing is assumed to occur at around a Reynold number of 800. At higher speed, turbulence starts to increase in the bearing. It manifests itself in heat generation within the bearing and in a significant increase in frictional losses.

TILTING-PAD JOURNAL BEARINGS

Tilting-pad journal bearings are selected for applications having light shaft loads due to their great ability to resist oil whirl vibration. However, these bearings can normally carry very high loads. Their pads can tilt to accommodate the forces developed in the oil film. Therefore, they can operate with an optimum thickness of the oil-film for a given application. This ability to operate over a wide range of loads is very useful for applications having high-speed gear reductions. The second advantage of tilting-pad journal bearings is their ability to accommodate shaft misalignment easily. These bearings should be used for high-speed rotors (which normally operate above the first critical speed) due to the advantages just listed and their dynamic stability.

Bearing preload is defined as the ratio of bearing assembly clearance to the machined clearance:

$$\text{Preload ratio} = \frac{C'}{C} = \frac{\text{Concentric pivot film thickness}}{\text{Machined clearance}} \qquad (16.1)$$

This is an important design criterion for tilting-pad bearings. A preload ratio of 0.5 to 1.0 provides stable operation due to the production of a converging wedge between the bearing journal and the bearing pads. The installed clearance of the bearing is C′. It depends on the radial position of the journal. For a given bearing, C is fixed. Figure 16.6 illustrates different preloading on two pads of a five-pad tilting-pad bearing. Pad 1 has a preload ratio less than 1, while Pad 2 has a preload ratio of 1.0. The solid line in Fig. 16.6 represents the position of the journal before applying the load. The dashed line represents the position of the journal after applying the load.

Pad 1 operates with a good converging wedge, while Pad 2 operates with a diverging film. This indicates that it is completely unloaded. Bearings operating with a preload ratio of 1 or higher will have some of their pads completely unloaded. This reduces the overall stiffness

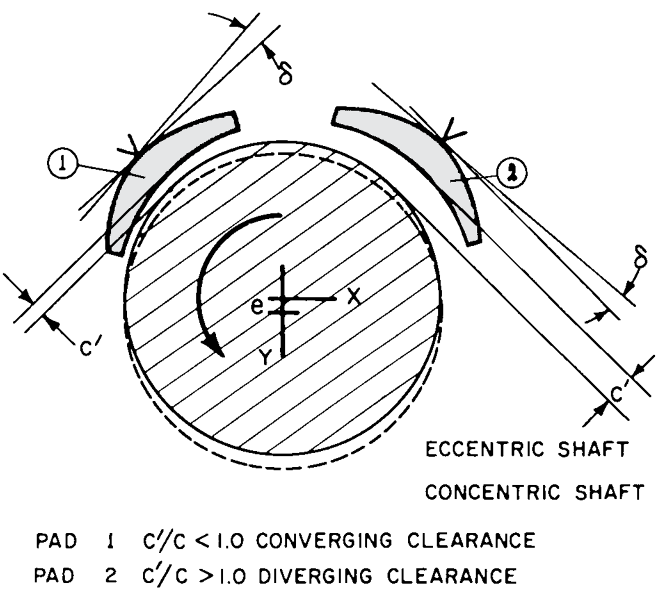


FIGURE 16.6 Tilting-pad bearing preload.

of the bearing and decreases its stability. Unloaded pads experience flutter, which leads to a phenomenon known as *leading-edge lockup.* In this situation, the pads would be forced against the shaft and maintained in this position by the friction of the shaft and the pad. Therefore, bearings should be designed with a preload, especially when the lubricant viscosity is low.

BEARING MATERIALS

Babbitt is the soft material in the stator of the bearing that faces the journal. It has excellent nonscoring characteristics and is outstanding for embedding dirt. However, it has low fatigue strength, especially at elevated temperatures and when its thickness is more than 0.038 cm (0.015 in). Babbitts will not be damaged by momentary rupture of the oil film. They will also minimize the damage to the journal in the event of a complete failure. Tin babbitts are preferred over lead-based material due to their higher corrosion resistance. They are also easier to bond to a steel shell.

The maximum design temperature of Babbitts is around 300°F (149°C). However, most applications are limited to 250°F (121°C). This metal tends to experience creep as the temperature increases. Creep normally forms ripples on the bearing surface. Tin babbitts experience creep from around 375°F (190°C) and for bearing loads below 200 psi (1.36 MPa) to 270°F (132°C) and for steady loads of 1000 psi (6.8 MPa). This range can be improved by using very thin layers of Babbitt as in automotive bearings.

BEARING AND SHAFT INSTABILITIES

Journal bearings encounter a serious form of instability known as *half-frequency whirl.* This phenomenon is caused by vibration characterized by rotation of the shaft center around the bearing center at a frequency of half the shaft rotational speed (Fig. 16.7).

Any increase in speed following this phenomenon will produce more violent vibration until eventual seizure occurs. Unlike a critical speed, the shaft cannot "pass through" this region. As the shaft speed increases, the frequency of instability remains at half the shaft speed. This problem occurs mainly at high speed in lightly loaded bearings. This problem can be predicted accurately and avoided by changing the design of the bearing. This problem does not occur in tilted-pad bearings. However, these bearings can become unstable due to the problem of pad flutter. The main cause of bearing failure is its inability to resist cyclic stresses. The severity charts in Fig. 16.8 show the level of vibration that can be tolerated by bearings.

THRUST BEARINGS

The main function of a thrust bearing is to resist any axial force applied to the rotor and maintain it in its position. Figure 16.9 illustrates three types of thrust bearings. The plain washer bearing is not normally used with continuous loads. Its applications are limited to thrust loads of very short duration, at standstill, or low speed. This type of bearings is used also for light loads [less than 50 psi (340 kPa)].

Thrust bearings designed to handle significant continuous loads require a fluid film between the bearing surface and the rotor. The tapered-land thrust bearing can match the load handled by the tilting-pad thrust bearing. However, tilting-pad thrust bearings are preferred for variable speed operation. The main reason for this is the ability of the pads

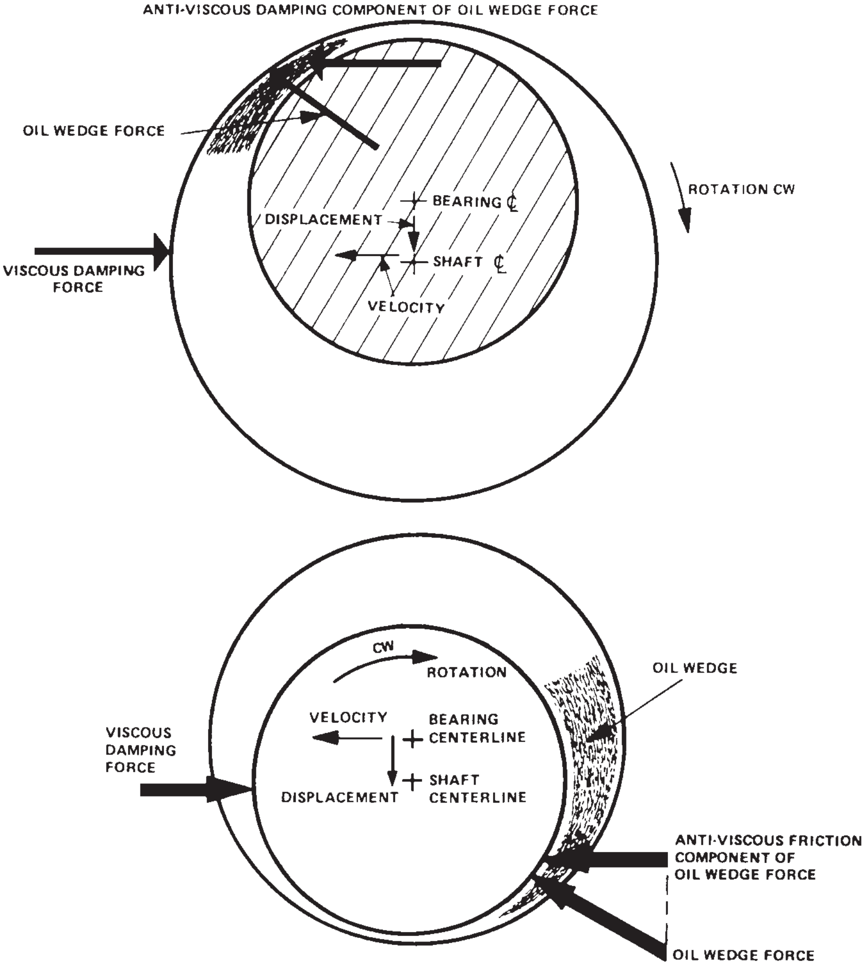
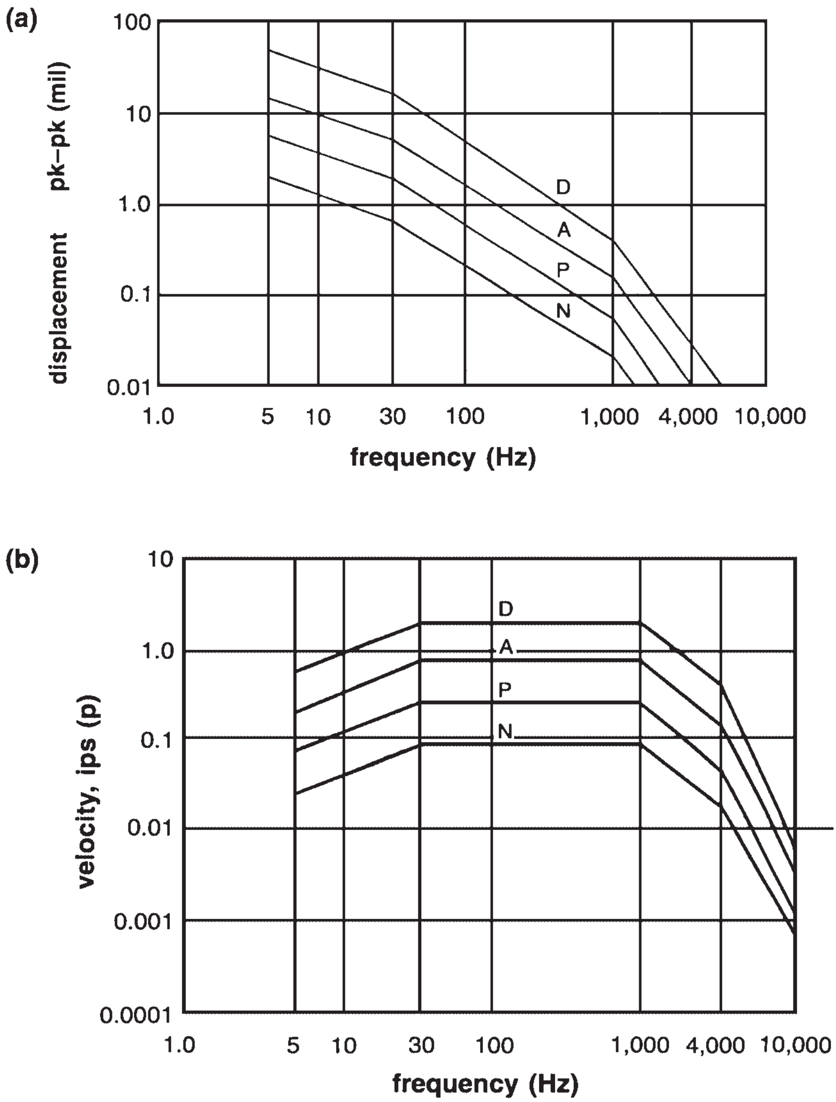


FIGURE 16.7 Oil whirl.

to pivot freely to form a suitable angle for lubrication over a wide speed range. The self-leveling feature equalizes the loads on the individual pads and allows the bearing to tolerate larger shaft misalignments. The main disadvantage of this bearing design is that it requires more axial space than nonequalizing thrust bearings.

Factors Affecting Thrust Bearing Design

Tests have proven that the load capacity of a thrust bearing is limited by the strength of the babbitt surface at the highest temperature in the bearing. The normal capacity of a steel-backed babbitted tilting-pad thrust bearing is around 250 to 500 psi (1700 to 3400 kPa). This capacity can be improved by maintaining the flatness of the pads and removing heat



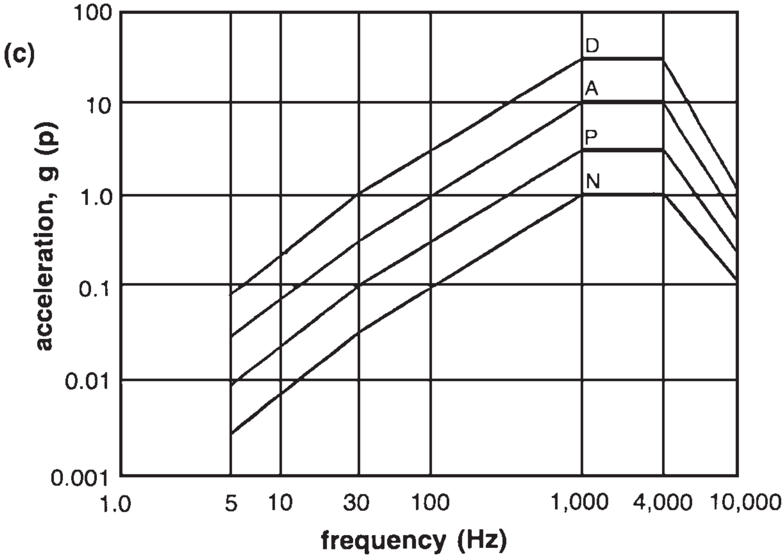
D = Dangerous, shutdown.

A = Abnormal, will deteriorate. Inspect as early as possible.

P = Problems, keep close watch.

N = Normal.

FIGURE 16.8 Severity charts: (*a*) displacement and (*b*) velocity.



D = Dangerous, shutdown.

A = Abnormal, will deteriorate. Inspect as early as possible.

P = Problems, keep close watch.

N = Normal.

FIGURE 16.8 (*Continued)* Severity charts: (*c*) acceleration.

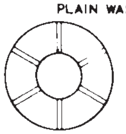
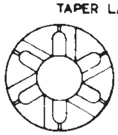
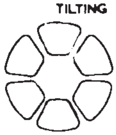
BEARING TYPE		LOAD CAPACITY	SUITABLE DIRECTION OF ROTATION	TOLERANCE OF CHANGING LOAD / SPEED	TOLERANCE OF MISALIGNMENT	SPACE REQUIRMENT
PLAIN WASHER 		POOR		GOOD	MODERATE	COMPACT
TAPER LAND 	BIDIRECTIONAL	MODERATE		POOR	POOR	COMPACT
	UNIDIRECTIONAL	GOOD		POOR	POOR	COMPACT
TILTING PAD 	BIDIRECTIONAL	GOOD		GOOD	GOOD	GREATER
	UNIDIRECTIONAL	GOOD		GOOD	GOOD	GREATER

FIGURE 16.9 Comparison of thrust-bearing types.

from the loaded area. The use of backing materials with proper thickness and high thermal conductivity can increase the maximum continuous thrust to more than 1000 psi (6800 kPa).

The use of backing material having high thermal conductivity, such as copper or bronze, allows the thickness of the Babbitt to be reduced to 0.01 to 0.03 in (0.025 to 0.076 cm). Thermocouples and resistive thermal detectors (RTD's) embedded in the bearing will signal distress when they are properly positioned. Temperature monitoring systems have proven to have a higher accuracy than axial position indicators, which tend to have problems with linearity at high temperatures.

Thrust Bearing Power Loss

The power consumed in a thrust bearing must be accurately predicted to determine the turbine efficiency and the requirements of the oil supply. Figure 16.10 illustrates the typical power consumption in a thrust bearing with shaft speed. The total power loss is around 0.8 to 1.0 percent of the total rated power of the machine. Newly tested vectored lube bearings show preliminary indications of reducing the power loss by 30 percent.

SEALS

Seals are critical components in turbomachinery, especially when the unit operates at high pressure and speed. The two categories of sealing systems between the rotor and the stator are (1) *noncontacting seals* and (2) *face seals.*

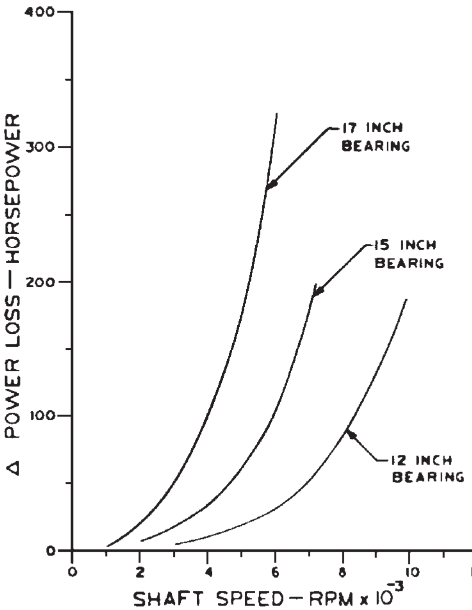


FIGURE 16.10 Difference in total power loss data—test minus catalog frictional losses versus shaft speed for 6 × 6 pad double-element thrust bearings.

Noncontacting Seals

Noncontacting seals are reliable and commonly used in high-speed turbomachinery. The two types of noncontacting seals (or clearance seals) are *labyrinth seals* and *ring seals.*

Labyrinth Seals. A labyrinth seal consists of a series of metallic circumferential strips that extend from the shaft or from the shaft housing to form a series of annular orifices. Labyrinth seals have higher leakage than clearance bushings, contact seals, or film riding seals. Thus, labyrinth seals are used in applications that can tolerate a small loss of efficiency. They are also used sometimes in conjunction with a primary seal. The advantages of labyrinth seals are:

- Simplicity
- Reliability
- Tolerance to impurities
- System adaptability
- Very low power consumption
- Flexibility of material selection
- Minimal effects on the rotor
- Reduction of reverse diffusion
- Ability to handle very high pressures
- Tolerance to large temperature variations

Their disadvantages are:

- Relatively high leakage
- Loss of efficiency
- Possible ingestion of impurities with resulting damage to other components such as bearings
- Possible clogging
- Inability to meet the seal standards of the Environmental Protection Agency (EPA)

Many modern machines are relying on other types of seals due to these disadvantages.

Labyrinth seals can easily be manufactured from conventional materials. Figure 16.11 illustrates some of the modern seals. The grooved seal shown in Fig 16.11(*b*) is tighter than the simple seal shown in Fig 16.11(*a*). Figures 16.11(*c*) and 16.11(*d*) show rotating labyrinth-type seals. Figure 16.11(*e*) shows a buffered, stepped labyrinth seal. This design is normally tighter than the one described earlier. Figure 16.11(*f*) shows a buffered-vented straight labyrinth seal. The pressure of the buffered gas is maintained at a higher value than the process gas, which can be under vacuum or above atmospheric conditions. The buffered gas produces a fluid barrier that seals the process gas. The eductor sucks the buffered gas and atmospheric air into a tank maintained under vacuum.

The matching stationary seal is normally made from soft material such as babbitt or bronze, while the rotating labyrinth lands are made of steel. This arrangement allows the seal to have minimal clearance. During operation, the lands can cut into the softer material without causing extensive damage to the seal. In a labyrinth seal, the high fluid pressure is converted into high velocity at the throats of the restrictions. The kinetic (velocity) energy is then dissipated into heat by turbulence in the chamber after each throat. The clearances of a large turbine is around 0.015 to 0.02 in (0.038 to 0.51 cm).

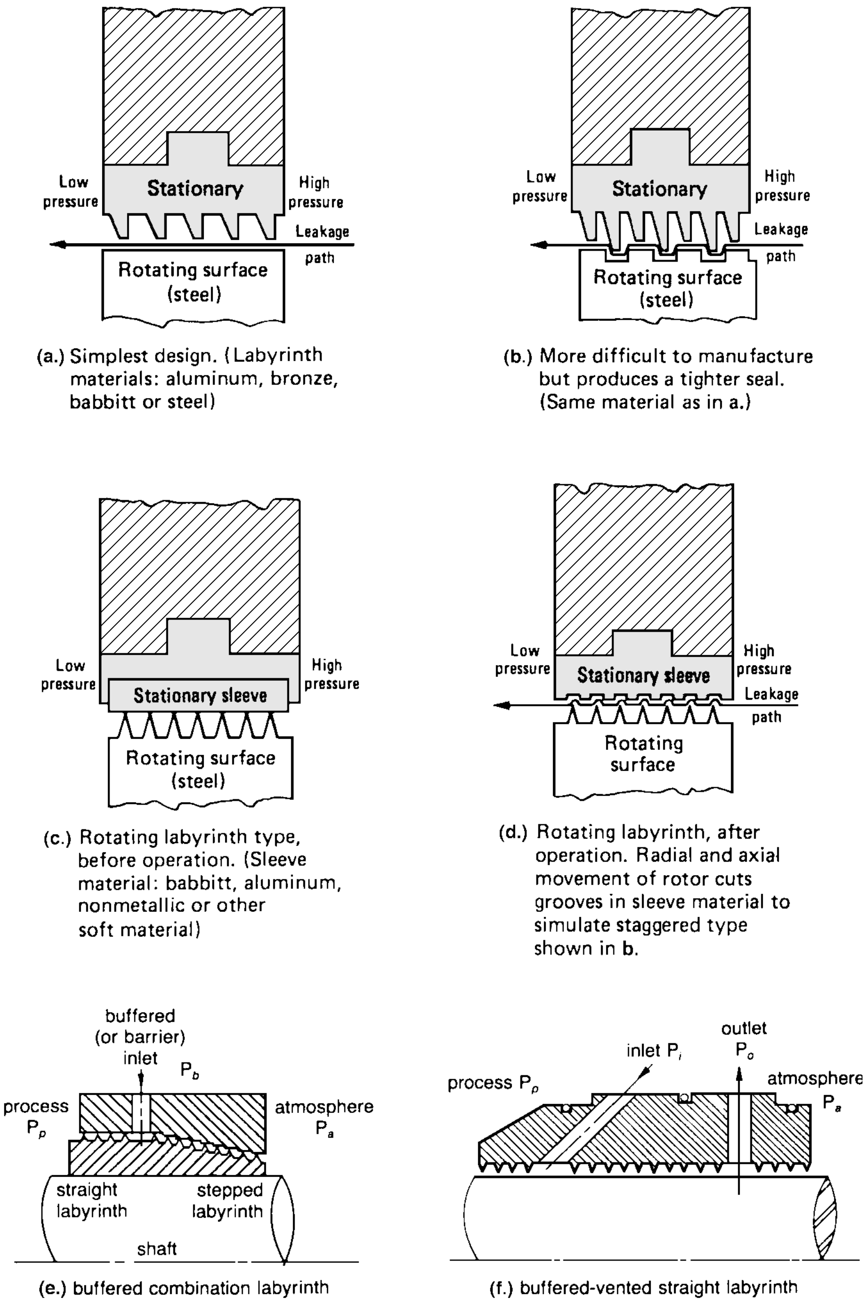


FIGURE 16.11 Various configurations of labyrinth seals.

Ring (Bushing) Seals. This seal consists of a series of sleeves having a small clearance around the shaft. The leakage across the seal is limited by the flow resistance. This design allows the shaft to expand axially when the temperature increases without affecting the integrity of the seal. The segmented and rigid types of this seal are shown in Figs. 16.12(*a*) and 16.12(*b*), respectively. This seal is ideal for high-speed rotating machinery due to the minimal contact between the stationary ring and the rotor.

The seal ring is normally made from babbitt-lined steel, bronze, or carbon. The main advantage of carbon is its self-lubricating properties. If the fluid is a gas, carbon seal rings

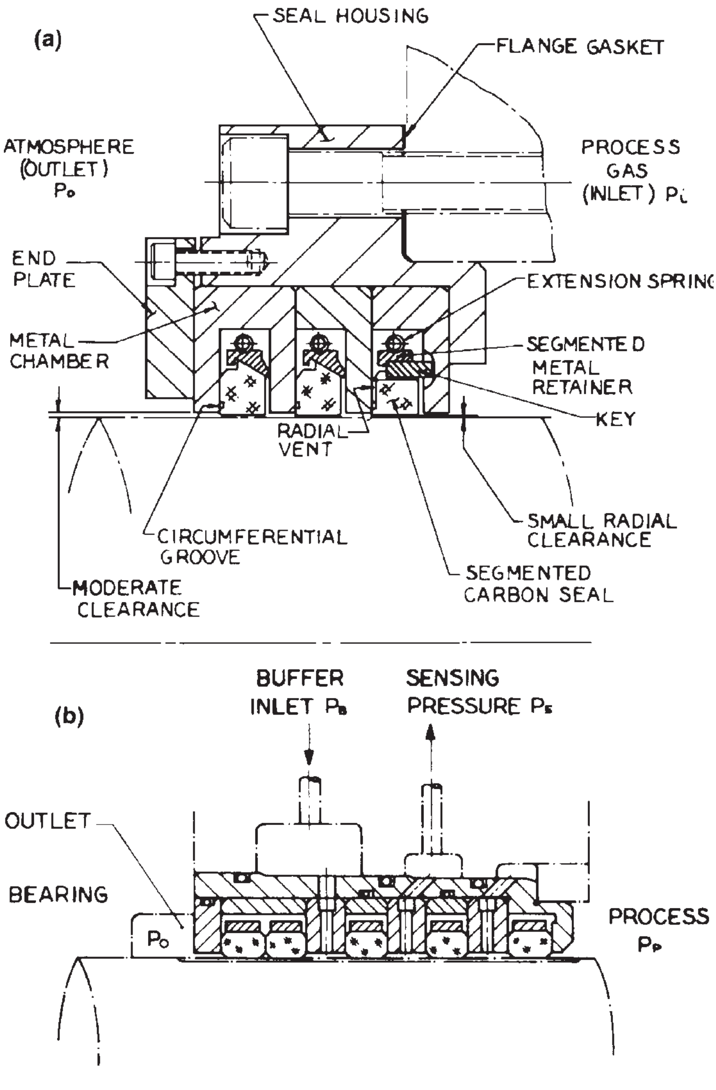


FIGURE 16.12 Floating-type restrictive ring seal.

should be used. Flow through the seal provides the cooling required. In some applications, seal rings are made from aluminum alloys or silver.

Mechanical (Face) Seals

The main purpose of a mechanical (face) seal is to prevent leakage. It consists of the following subcomponents:

- A stationary seal ring mounted around the shaft known as the *stator* of the seal
- A rotating seal ring mounted on the shaft known as the *rotor* of the seal
- Springs to push the rotating ring against the stationary ring
- Static seals (o-rings)

The sealing surfaces of the rotor against the stator are normally in a plane perpendicular to the shaft. The forces that hold these surfaces together are parallel to the shaft. Figure 16.13

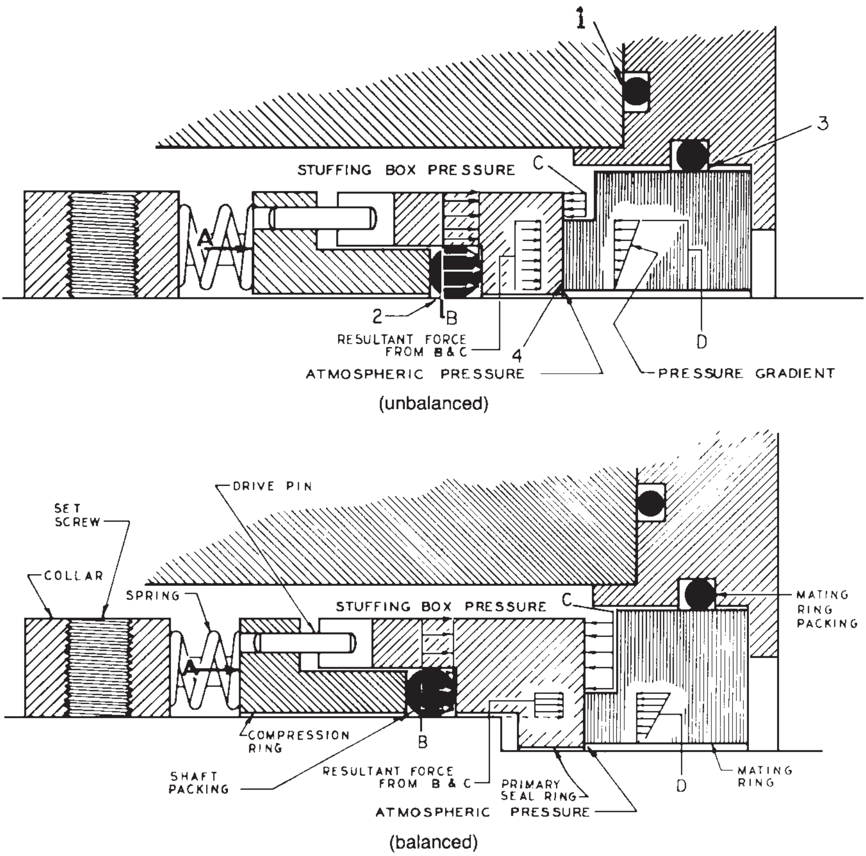


FIGURE 16.13 Unbalanced seal and balanced seal with step in shaft.

illustrates the four sealing points that must be sealed to ensure adequate operation of the seal:

1. The stuffing-box face
2. Leakage along the shaft
3. The mating ring in the gland plate
4. The dynamic faces (rotary to stationary)

The basic units of the seal (Fig. 16.14) are the *seal head* and the *seal seat.* The seal head unit includes the housing, the end-face member, and the spring assembly. The seal seat is the member that mates the seal head. The faces of the seal head and seal seat are lapped to ensure a flatness of $3 \times 10^{-6} - 15 \times 10^{-6}$ in ($8 \times 10^{-6} - 38 \times 10^{-6}$ cm). The head or the seat must rotate, while the other remains stationary. During normal operation, the sealing surfaces are kept closed by the hydraulic pressure. The spring is only needed to close the sealing surfaces when the hydraulic pressure is lost. The degree of *seal balance* (Fig. 16.15) determines the load on the sealing area. A completely balanced seal will only have the spring force acting on the sealing surfaces (i.e., there is no net hydraulic pressure on the sealing surfaces).

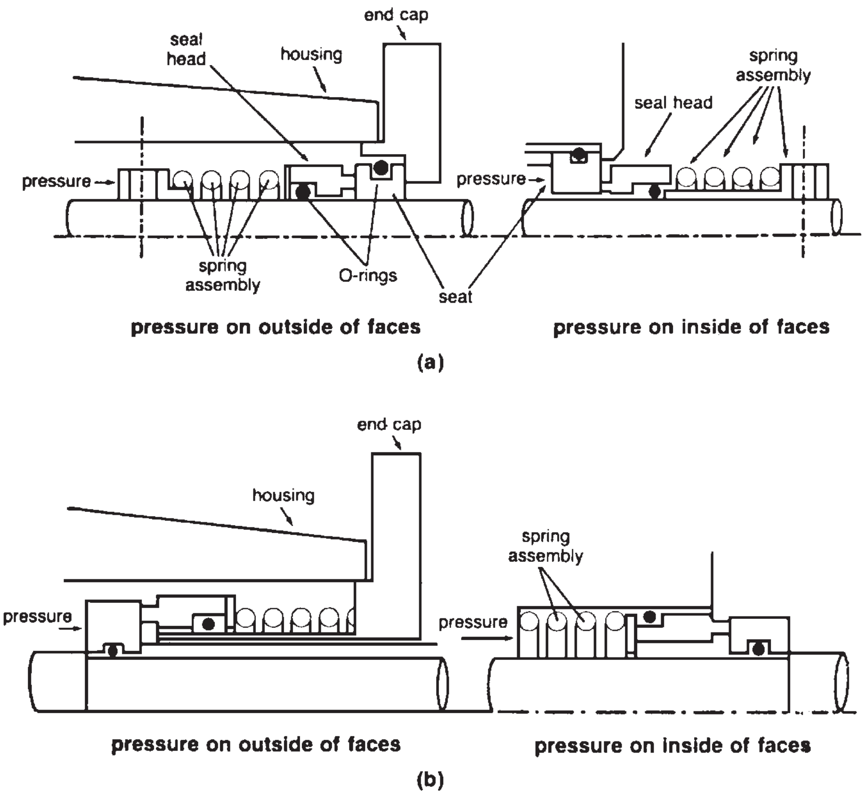


FIGURE 16.14 (*a*) Rotating and (*b*) stationary seal heads.

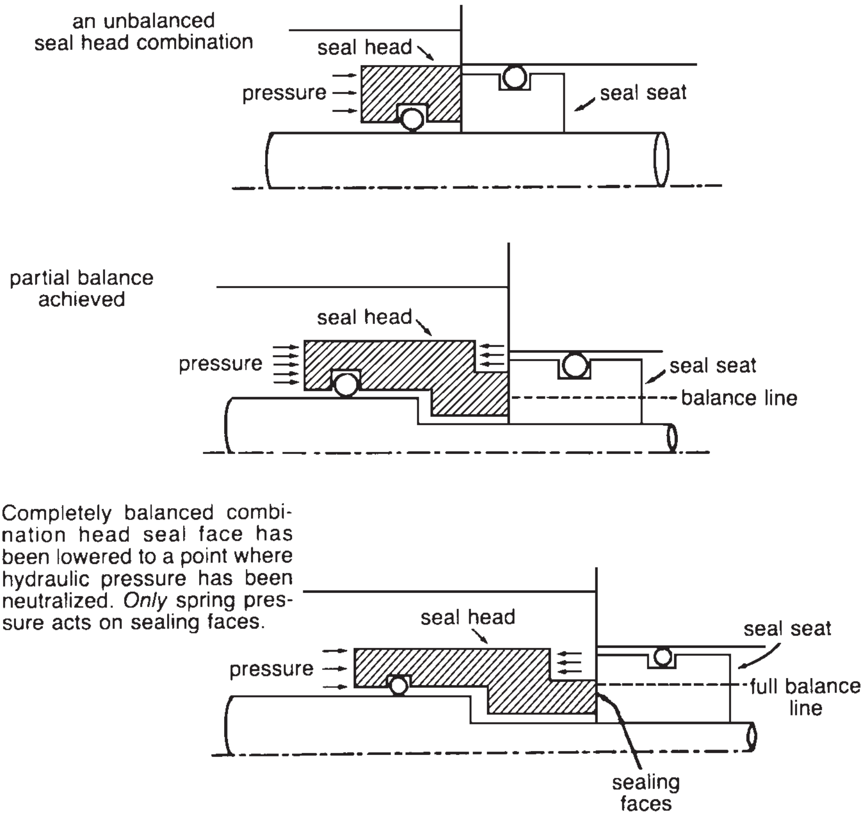


FIGURE 16.15 The seal balance concept.

During the last decade, magnetic seals (Fig. 16.16) have proven to be reliable under severe operating conditions for a variety of fluids. They use magnetic force to produce a face loading. Their advantages are that they are compact, relatively lighter, provide an even sealing force, and are easy to assemble. The two groups of shaft seals are:

- *Pusher-type seal.* It includes o-ring, v-ring, U-cup, and wedge configuration (Fig. 16.17).
- *Bellow-type seals.* They form a static seal between themselves and the shaft.

The two main elements of a mechanical contact shaft seal (Fig. 16.18) are: the oil-to-pressure-gas seal and the oil-to-uncontaminated-seal-oil-drain seal known as *breakdown bushing.* A buffer gas is injected at a port inboard of the seal. During shutdown, the carbon ring remains tightly sandwiched between the rotating seal ring and the stationary sleeve to prevent gas in the compressor from leaking out when the seal oil is not applied.

During operation, the seal oil is maintained at a pressure of 35 to 50 psi (238 to 340 kPa) higher than the process gas. The seal oil enters from the top of the seal and fills the seal cavity completely. A small oil flow is forced across the seal faces of the carbon ring to provide lubrication and cooling for the seal. The oil that crossed the seal faces contacts the process gas. It is called *contaminated oil.* The majority of the oil flows out from the uncontaminated seal oil

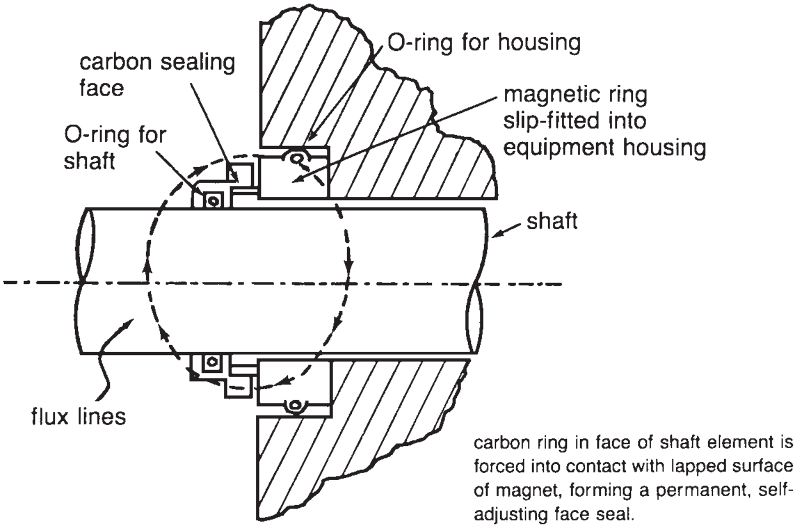


FIGURE 16.16 Simple magnetic-type seal.

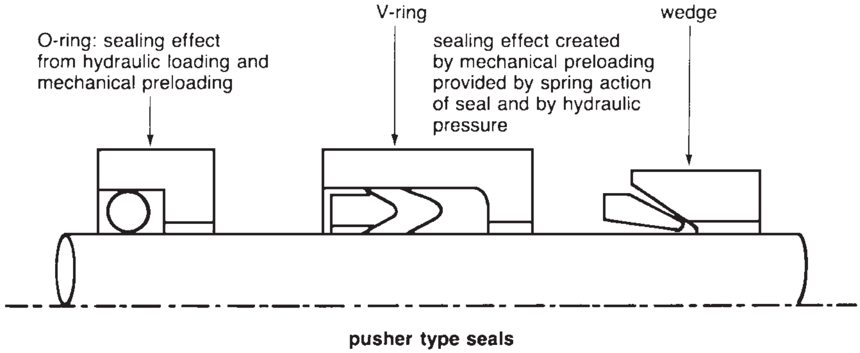
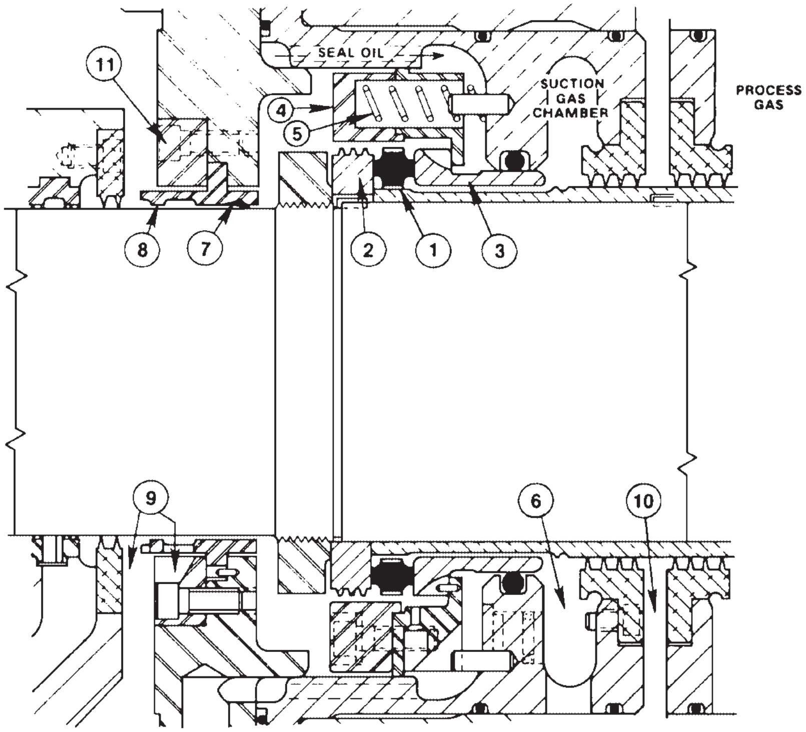


FIGURE 16.17 Various types of shaft sealing elements.

drain line (item 9). The contaminated oil leaves through the drain (item 6) to be purified in the degasifier. In some applications, the bearing oil is combined with the uncontaminated seal oil. However, a separate system for the bearing oil will increase the life of the bearings.

SEAL SYSTEMS

Modern sealing systems have become more sophisticated to meet recent government regulations. Figure 16.19 illustrates a simple seal having a buffered gas and an eductor. The buffer gas pressure must be subatmospheric. Problems have occurred with these systems due to the low capacity of the eductor and variations in the buffer gas pressure. Figure 16.20 illustrates a modern complex seal that incorporates three different types of seals to provide the most effective sealing arrangement. The labyrinth seal prevents the polymers in the



1. ROTATING CARBON RING
2. ROTATING SEAL RING
3. STATIONARY SLEEVE
4. SPRING RETAINER
5. SPRING
6. GAS AND CONTAMINATED OIL DRAIN
7. FLOATING BABBITT-FACED STEEL RING
8. SEAL WIPER RING
9. SEAL OIL DRAIN LINE
10. BUFFER GAS INJECTION PORT
11. BYPASS ORIFICE

FIGURE 16.18 Mechanical contact shaft seal.

process gas from clogging the seal rings. Following the labyrinth seal there are two segmented circumferential contact seals and four segmented restrictive-ring seals. This combination makes the primary seal. Four circumferential-segmented seal rings follow the primary seal. A buffer gas is injected at the first set of circumferential contact seals. An eductor is also installed at the rear circumferential seals. Thus, this assembly is very effective in providing a tight seal in most applications.

REFERENCE

1. Boyce, Meheran P., *Gas Turbine Engineering Handbook,* Gulf Publishing Company, Houston, Tex., 1982, reprinted July 1995.

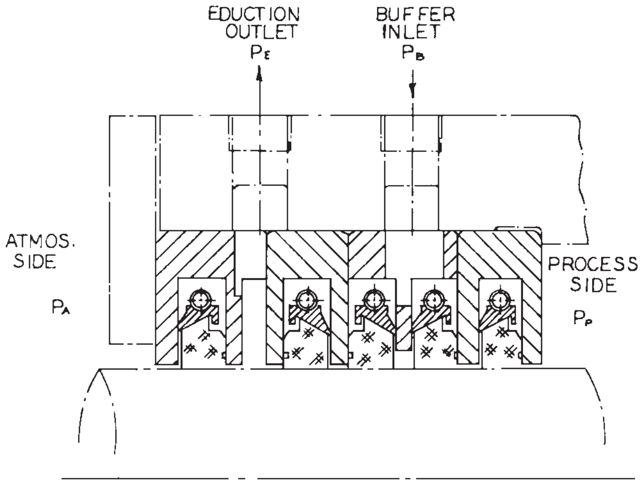


FIGURE 16.19 Restrictive ring seal system with both buffer and education cavities.

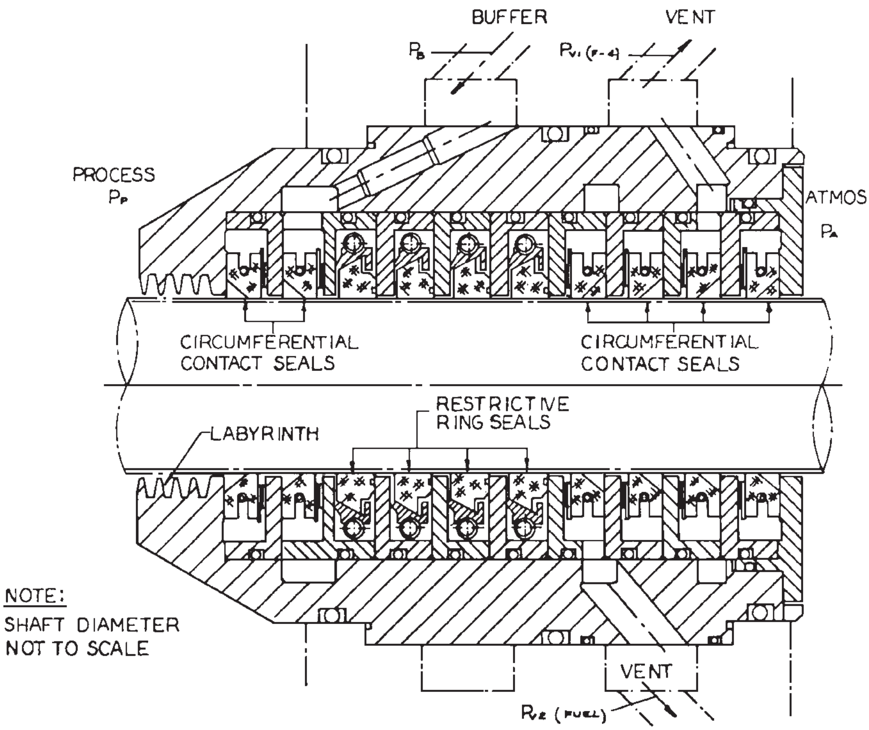


FIGURE 16.20 Multiple combination segmented gas seal system.